

Performance of Four-Wheel-Steering Vehicles  
in Lane Change Maneuvers

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**Summary**

This study addresses the performance of four-wheel-steering vehicles in high-speed lane change maneuvers. We compare the steering commands of an experienced driver in executing high-speed lane change maneuvers in road tests with those determined via solving suitably formulated optimization problems. It turns out that the optimal control determined is qualitatively comparable to the steering commands used by experienced drivers in road tests. Hence, we can analytically compare the performance of an experienced driver in executing lane change maneuvers using different vehicles. For a representative high-speed lane change maneuver, our study revealed that, in the hands of an experienced driver, the performance benefit achievable with four-wheel-steering vehicles (using either an open-loop or a closed-loop control algorithm) is not significant relative to that achievable with a two-wheel-steering vehicle. This conclusion confirms road-test results obtained with two production four-wheel-steering vehicles. The consistency between the road-test results and the conclusion obtained from the "optimal control" approach indicates the potential of the proposed methodology as a tool in evaluating the performance of driver/4 designs in safety-related maneuvers.

**Keywords:** Four-wheel-steering, Lane change maneuvers, Obstacle avoidance.

## introduction

Four-wheel-steering (4WS) systems for passenger vehicles have been actively studied recently.<sup>1</sup> The performance of these systems depends largely on how the rear wheels are controlled as a function of the forward speed of the vehicle, the steering angle, and other vehicle states. These open-loop or closed-loop controllers are usually designed to improve (1) vehicle maneuverability at low speed, and (2) straight-line stability (in cross wind) at high speed. However, the performance of four-wheel-steering vehicles in collision avoidance maneuvers has not been adequately evaluated.

This paper compares the performance of 2WS and 4WS vehicles in high-speed lane change maneuvers. To this end, the dynamical and kinematical models of the vehicle are first derived, together with the steering actuator dynamics. Based upon these models, open-loop and closed-loop 4WS algorithms are designed. Next, we compare the steering command of an experienced driver in executing a high-speed lane change maneuver with that determined via solving an optimization problem. The optimization problem is formulated with a cost functional that includes the lane change time as well as the desired conditions of the vehicle both during and at the end of the lane change maneuver. It turns out that the optimal control determined is qualitatively comparable to the steering commands used by experienced drivers in road tests. Accordingly, we can relate the optimal result obtained with either a 2WS or a 4WS vehicle to the corresponding performance achievable by an experienced driver, using the same vehicle to execute an identical lane change maneuver. In this way, we can analytically determine how well experienced drivers can execute collision avoidance maneuvers using either a 2WS or a 4WS vehicle.

## Vehicle Dynamics Model

Consider a vehicle moving over a flat and level road surface (Fig. 1). When the forward speed,  $U$ , is kept constant, this vehicle model has two-degrees-of-freedom represented by the side velocity  $v$  and the yaw-rate  $r$ . The side velocity  $v$ , defined at the vehicle's center of gravity (e.g.), is that component of the vehicle velocity vector that is perpendicular to its axis. The cornering forces acting on the front and rear axles are denoted by  $F_f$  and  $F_r$ , respectively. Apart from these forces, there are the relatively small aligning torques, camber angle effects, etc. that are neglected in our

study. Accordingly, the equations of motion are

$$M_s(Ur + \dot{v}) = F_f - F_r, \quad (1)$$

$$I_{zz}\dot{r} = aF_f - bF_r, \quad (2)$$

where  $a$  and  $b$  define the location of the vehicle's e.g. between the axles, and  $M_s$  and  $I_{zz}$  denote the mass and the yaw moment of inertia of the vehicle about the z-axis, respectively. The pitch and the roll dynamics of a vehicle do not significantly affect its directional behavior. They are neglected in this simplified analysis.

The lateral force produced by a tire is proportional to the tire slip angle, which is the angle between the direction of motion and the center-plane of the tire. Accordingly, we have

$$F_f = -2C_{\alpha_f} \alpha_f, \quad F_r = -2C_{\alpha_r} \alpha_r, \quad (3)$$

where  $C_{\alpha_f}$  and  $C_{\alpha_r}$  denote the cornering stiffnesses of each front and rear tire, respectively. The cornering stiffness of a tire is the ratio of the produced lateral (or cornering) force by its slip angle. The slip angles  $\alpha_f$  and  $\alpha_r$  are given respectively by the following kinematical relations

$$\alpha_f = \frac{v + ar}{u} - \delta_f, \quad \alpha_r = \frac{v - br}{u} - \delta_r, \quad (4)$$

where  $\delta_f$  and  $\delta_r$  denote the front and rear tire angles, respectively. Combining equations (1) through (4), we have

$$\begin{aligned} I_{zz}\dot{r} + \frac{2(a^2C_{\alpha_f} + b^2C_{\alpha_r})}{U}r + 2(aC_{\alpha_f} - bC_{\alpha_r})\frac{v}{U} &= 2aC_{\alpha_f}\delta_f - 2bC_{\alpha_r}\delta_r, \\ M_s\dot{v} + (M_su + \frac{2(aC_{\alpha_f} - bC_{\alpha_r})}{U})v + 2(C_{\alpha_f} + C_{\alpha_r})\frac{v}{U} &= 2C_{\alpha_f}\delta_f + 2C_{\alpha_r}\delta_r. \end{aligned} \quad (5)$$

In our study, the above model is augmented with the following first-order actuator dynamic models

$$\tau_f\dot{\delta}_f + \delta_f = \delta_{fc}, \quad \tau_r\dot{\delta}_r + \delta_r = \delta_{rc}, \quad (6)$$

Here  $\delta_{fc}$  and  $\delta_{rc}$  are commands to the front and rear actuators, respectively. In (6),  $\tau_f$  and  $\tau_r$  are the time constants of the front and rear actuators, respectively. In our

study, we assumed that the bandwidth of these actuators, for evasive maneuver scenarios (vehicle speed  $\geq 100$  /h, lateral acceleration  $\geq 0.6g$ , and steering wheel speed  $\geq 500$  cdeg/sec) is 4 Hz.

in addition to these dynamical equations, the following kinematical relations are used to compute the resultant trajectory of the vehicle:

$$\begin{aligned}\dot{\psi} &= r, \\ \dot{x} &= U \cos \psi - v \sin \psi, \\ \dot{y} &= U \sin \psi + v \cos \psi.\end{aligned}\tag{7}$$

With reference to Fig. 1,  $(x,y)$  is the rectilinear coordinate of the vehicle's e.g. relative to an arbitrary reference, and  $\psi$  is the angle the vehicle's axis made with the  $\hat{x}$ -axis, positive in the clockwise direction (Fig. 5).

Parameters for typical passenger vehicles may be found in Ref. 2. In Ref. 2, we use  $a = 1.2$  m,  $b = 1.6$  m,  $I_{zz} = 2200$  kg-m<sup>2</sup>,  $M_s = 1700$  kg,  $C_{\alpha_f} = 960$  N/deg, and  $C_{\alpha_r} = 1100$  N/deg. The validity of the above described linear vehicle model begins to deteriorate in maneuvers that exhibit lateral acceleration in excess of 0.3 g's. Unfortunately, high-speed lane change maneuvers, to be studied here, are characterized by lateral acceleration levels as high as 0.6 g's. However, the situation is mitigated somewhat by the fact that these high-g conditions only lasted for a short time. Hence, we will continue to use this model in our study, but will make adjustments to the cornering stiffnesses of the front and rear tires so that responses obtained using this model match reasonably well with those found from field tests at up to 0.6 g's. Obviously, this linear vehicle model cannot be used to study emergency maneuvers that lead to spin-out and/or roll-over. To do that, a nonlinear model that includes tire saturation effect must be used (Ref. 3).

Based upon the above described model, we can design 4WS algorithms to augment the lateral stability of the vehicle at high speed. To this end, we consider the following representative open-loop and closed-loop 4WS algorithms.

### Open-loop and Closed-loop 4WS Algorithms

4WSN Algorithm This is an open-loop algorithm suggested by Nissan Motor Company. Using a vehicle model, a speed-dependent ratio between the rear and front wheels

is computed in order to achieve zero steady-state side velocity:  $\delta_{rc}/\delta_{fc} = K_N(U)$ , where  $U$  is the forward speed of the vehicle. The function  $K_N$  can be easily computed by equating the terms  $\dot{v}$ ,  $\dot{r}$ , and  $v$  in (5) to zero, and solving for the resultant ratio of rear to front steering angles. The results indicate that the rear wheels are steered out-of-phase with respect to the front wheels at low speed, and in-phase at high speed. For example,  $\delta_{rc}/\delta_{fc} = -0.39$ ,  $-10.00$ , and  $-1.036$  at 40, 60, and 100 km/h, respectively.

At high speed, in-phase steering of the rear wheels generates lateral forces that counteract with those produced at the front, and the response time of the vehicle's yaw rate will deteriorate. Additionally, while the side velocity approaches zero in the steady state, its transient value is larger and in the opposite direction relative to that produced by a 2WS vehicle. This is disconcerting to the driver. These problems can be partially alleviated by delaying the execution of the rear wheel command by a short time:<sup>1</sup>

$$\delta_{rc}(t) = K_N(U) \delta_{fc}(t - \tau_D) . \quad (8)$$

Here,  $\tau_D$  is the delay time and is iteratively determined to be 0.08 seconds. An electro-hydraulic servo system in conjunction with a microcomputer were used to implement such an algorithm in Ref. 3. The sensors used in the system include the front and rear steering angle sensors and a vehicle velocity sensor. Numerous other simple open-loop algorithms have also been suggested.<sup>4</sup>

**4WSY Algorithm** This is a simple closed-loop algorithm with feed-through of the front steering command and feedback of the vehicle's yaw-rate

$$\delta_{rc} = -K_r(U) [Y_G(U) \delta_{fc} - \left( \frac{1 + \tau_1 s}{1 + \tau_2 s} \right) r] \quad (9)$$

The function  $Y_G(U)$  is a speed-dependent yaw velocity gain of the deg2WS vehicle (Fig. 3). The  $(1 + \tau_1 s)$  is a "lead" term ( $\tau_1 = 0.01$  seconds), and  $\tau_2$  is the time constant of a low-pass filter that is used to "clean up" the noisy signal from a yaw rate gyroscope ( $\tau_2 \approx 0.005$  seconds). The speed-dependent feedback gain  $K_r$  is selected as a compromise among the transient responses of the vehicle's yaw-rate, lateral acceleration, and steering rate (driver workload). For simplicity, we used a constant value of  $K_r = 2.5$  /(deg/sec) at all vehicle speeds. In addition to the needs of having sensors for the steering angles and vehicle velocity, this closed-loop algorithm also requires the use of a gyroscope to measure the vehicle's yaw rate. Other closed-loop algorithms have also been proposed.<sup>5</sup>

## Steady-state and Transient Performances of 2WS and 4WS Vehicles

Two 4WS algorithms, 4WSN and 4WSY, were designed and studied. The lateral acceleration and yaw velocity gains for the 2WS, 4WSN, and 4WSY vehicles as functions of vehicle speed are compared in Figs. 2 and 3, respectively. These gains are defined as the steady-state values of the vehicle's lateral acceleration (at the vehicle's e.g. ) and yaw rate, per each degree of front tire angle, respectively. The ratio of the vehicle's steady-state lateral acceleration to the steering wheel angle is also called its steering sensitivity or control gain. Note that the steady-state gains for the 2WS and 4WSY vehicles are identical because the 4WSY algorithm, like the 2WS vehicle, produces no rear-steering angle in the steady state. Those for the 4WSN vehicle are different because of the nonzero steady state rear steering angle.

The lateral acceleration gains  $A_G(U)$  of both the 2WS (and 4WSY) and 4WSN vehicles increase with vehicle speed, approaching a finite value,  $1/1\{$ , when the vehicle speed becomes very high. Here  $K$  is the understeer coefficient of the vehicle. From Fig. 2, we see that steering the rear wheels of a vehicle in-phase with the front ones causes a drop in the lateral acceleration gain (or an increase in the understeer coefficient). This is undesirable since the driver will have to turn through a larger steering angle (relative to a 2WS vehicle) in order to generate the same level of lateral acceleration.

The yaw velocity gains  $Y_G(U)$  of both the 2WS (and the 4WSY) and 4WSN vehicles increase and then decrease with the vehicle speed (Fig. 3). These gains reach their maximum values at their characteristic speeds ( $\approx 100$  and  $40$  km/h for the 2WS and 4WSN car, respectively). Since the value of a vehicle's lateral acceleration gain  $\approx 1/2K$  at its characteristic speed, the approximate understeer coefficients for the 2WS (and 4WSY) and 4WSN vehicles are  $1.9$  and  $5.2$  deg/g respectively (Fig. 2). At speed higher than  $60$  km/h, the yaw rate gain of the 4WSN vehicle is lower than that of the 2WS vehicle due to the in-phase steering of the front and rear wheels.

Transient responses of the 2WS, 4WSN, and 4WSY are depicted in Figs. 4 and 5. In Fig. 4, the yaw rate responses of these vehicles to a "step" front wheel command ( $\delta_{fc}$ ) at  $120$  km/h are compared. Since a true steering step is physically impossible, the steering command is ramped to its steady-state value ( $\approx 0.5$  deg) over a time period of  $0.15$  seconds. Relative to the time response of the 2WS vehicle, those associated with the 4WSN and 4WSY are better damped with smaller overshoot and shorter settling

time. The percent overshoot ( $M_p$ ) and 90% rise-time ( $T_r$ ) of these vehicles' yaw-rate and lateral acceleration responses at a forward speed of 120 km/h are tabulated in Table 1. From that table, we observe that both the 4WSN and 4WSY vehicle can provide an improvement in the directional stability of the vehicle. The yaw rate response time of the 4WSN vehicle is very close to that of the 2WS vehicle while that of the 4WSY vehicle is significantly better.

Table 1. Vehicle Performance at 120 km/h

Criterion	2WS	4WSN	4WSY
$M_p$ (%) [ $r$ ]	20	6	2
$T_r$ (sec) [ $r$ ]	0.25	0.15	0.15
$M_p$ (%) [ $a_{yy}$ ]	3	4	0
$T_r$ (sec) [ $a_{yy}$ ]	0.48	0.23	0.45

### Performance of Driver-Vehicle System in Lane Change Maneuvers

The performance of a driver-vehicle system in collision avoidance maneuvers is difficult to evaluate because one must take both the vehicle's directional characteristics, as well as the limitations of driver responses, into consideration. Of the multitude of collision avoidance scenarios that one can envision, we have constructed for our study the following "representative" scenario (Fig. 5 illustrates the scenario considered) which a driver might encounter at any time on the highway. Other collision avoidance scenarios that had been studied in the literatures can be found in Refs. 3, 7, and 8.

As depicted in Fig. 5, a vehicle is traveling at a constant speed on a straight two-lane roadway when an object clashes into the vehicle's path and stop. Driver reactions when faced with such an emergency typically involved first a delay time, then an application of the brake, and finally the turning of the steering wheel in attempting to avoid the obstacle. A reaction time, on the order of 0.3 to 0.4 seconds between the appearance of the obstacle and the time the driver begins to respond has been quoted in the literature.<sup>G-9,10</sup> Braking is commonly used in these situations to decelerate the vehicle. This by itself is rarely sufficient, and the vehicle must be quickly and skillfully steered to a neighboring lane to avoid the obstacle. At times, drivers who are surprised

by the sudden appearance of an obstacle turn the steering wheel so abruptly that their vehicles go out of control or collide with cars in neighboring lanes.

In our study, we focused attention on the performance of a driver-vehicle system associated with the “steer-to-avoid” strategy rather than the “brake-and-stop” strategy. Road tests of constant-speed lane change maneuvers were conducted using both experienced and inexperienced drivers.<sup>6,9</sup> Typical time histories of the steering wheel excursions recorded in road tests are given in Fig. 6. As depicted, the initial steering commands generated by both driver groups are surprisingly similar. The maximum steering angles and the steering rates for both driver groups are on the order of 200 degrees and 800 /sec, respectively. This maximum steering angle generally corresponds to a steering excursion that can be turned with both hands on the wheel (Ref. 9). The initial steering command must be followed by an almost “equal- and- opposite” steering in order to arrest the diverging vehicle’s heading angle and return it back to the desired straight-ahead heading. Again, there are only minor differences between the recorded steering commands from the two driver groups.

In this study, we conjecture that driver steering commands in a lane change maneuver consist of a reflexive phase followed by a regulatory phase (Fig. 6). Since most drivers are aware of the importance of executing the steering command as quickly as possible, their dependency on visual feedback during the reflexive phase of a lane change maneuver will be relatively low. Instead, based on their estimates of the vehicle speed and lateral displacement needed to avoid the obstacle, a series of well-learned steering commands will be executed in “open-loop.” At the end of the reflexive phase, the vehicle has been displaced approximately the desired lateral displacement and has almost the desired straight-ahead heading. However, additional steering adjustments are still needed to “zero” out small residuals in the vehicle’s yaw rate, side velocity, and heading angle in the “Regulatory” phase. In this phase, an experienced driver will use his estimate of the vehicle positioning with respect to the roadway to generate small closed-loop steering adjustments to nullify the residual rates. Inexperienced drivers tend to “over correct” in the regulatory phase with large and oscillatory steering commands, leading to a relatively long “settling time.” The distinction between the open-loop and closed-loop phases of lane change maneuvers has also been mentioned in Ref. 10.

The above described results were obtained with a conventional 2WS vehicle. The



performance of a driver-4WS vehicle system in a similar lane change maneuver might be different. To study that, a 8-dof vehicle model was coupled to a driver model in Ref. 8. The lane change performance of 2WS and 4WS vehicles (with both open-loop and closed-loop 4WS algorithms) are then analytically compared. Results obtained indicate that a 4WS vehicle (with a closed-loop yaw rate-feedback algorithm) has the best lane change performance among the three vehicle configurations studied.

A drawback of the approach taken in Ref. 8 is the fact that the driver model used was one developed for conventional 2WS vehicles. To use it to study and predict the performance of a driver-4WS vehicle systems might not be suitable. This shortcoming also points to the urgent need to develop reliable driver models that can be used with 4WS vehicles (Refs. 10-12). Before these reliable driver models are available, we consider the following approach to analytically study lane change maneuvers.

### Optimal Vehicle Control in A Lane Change Maneuver

The steering control used, by an experienced driver in the reflexive phase of a lane change maneuver can also be estimated via solving a dynamical optimization problem. RefTo this end, we must augment the dynamical and kinematical equations of the vehicle (5-7) with a driver neuromuscular model. This is typically a critically damped second-order dynamic model, but for simplicity, we use the following first-order model in this study:

$$\tau_m \delta_{fc} + \delta_{fc} = G_{T_s}^{\delta_f} T_s / N_S \delta_{driver} . \quad (10)$$

Here,  $G_{T_s}^{\delta_f}$  (radian/Nm) represents the steering angle to wheel torque gain,  $T_s$  (Nm) is the steering torque command from the driver, and  $N_S$  is the ratio between the angular excursions of the steering wheel to the front tires ( $N_S = 15$ ). Collectively, the term on the right-hand-side of (10),  $\delta_{driver}$ , represents the steering command from the driver to the front steering actuator. The bandwidth of the driver response depends in part on the available power assist, and is assumed to be 2 Hz in our study.<sup>12</sup>

A dynamical optimization problem can be formulated as follows: Determine the control time history of  $\delta_{driver}$  to bring the vehicle from its initial to final state while minimizing a cost functional  $J$ . Here, the "states" of the driver-vehicle system consist of yaw rate ( $r$ ), side velocity ( $v$ ), front tire angle ( $\delta_f$ ), front tire angle command ( $\delta_{fc}$ ),

heading angle ( $\psi$ ), lateral displacement ( $y$ ), and the longitudinal displacement ( $x$ ). Immediately before the driver's response, the vehicle is in its straight ahead cruising condition. Hence, the initial conditions of all the variables are zero. To avoid the obstacle, the vehicle must be displaced a lateral distance  $D$  as soon as possible. Hence, the end condition of  $y$  is  $D$ . Additionally, it is desirable to return the vehicle's yaw rate and side velocity back to zero at the end time.

A candidate of the cost functional  $J$  is:

$$J = W_T \left( \frac{T}{T_N} \right) + \frac{1}{2} \left[ \left( \frac{\psi}{\psi_N} \right)^2 + \left( \frac{\delta_f}{\delta_{fN}} \right)^2 + \left( \frac{\delta_{fc}}{\delta_{fcN}} \right)^2 \right]_F + \frac{1}{2} \int_0^T \left[ \left( \frac{a_{yy}}{a_{yyN}} \right)^2 + \left( \frac{\dot{\delta}_{fc}}{\dot{\delta}_{fcN}} \right)^2 \right] \left( \frac{dt}{T_N} \right). \quad (11)$$

The first term in (11) accounts for the driver's overwhelming desire to complete the lane change as quickly as possible. The end-time of the maneuver,  $T$ , has been normalized with respect to a nominal time  $T_N (= 1 \text{ second})$  to make it dimensionless. The parameter  $W_T$  determines the relative importance of the maneuver time versus vehicle conditions both at the end time and during the course of the lane change maneuver (to be described next).

The subscript "F" in the second component of  $J$  denotes the conditions of  $\psi$ ,  $\delta_f$ , and  $\delta_{fc}$  at  $T$ . Looking at (11), it is clear that we wish to return  $\psi$ ,  $\delta_f$ , and  $\delta_{fc}$  back to "zero" as closely as possible at the end time. Again, these variables have been normalized using their "nominally acceptable" values. The nominally acceptable values of  $\psi$ ,  $\delta_f$ , and  $\delta_{fc}$  at the end-time are two degrees. If the vehicle's heading angle at the end time is below two degrees, the contribution of the term  $[\frac{\psi}{\psi_N}]^2$  in  $J$  becomes small, and vice versa. Additionally, the steering control must also ensure that the lateral acceleration experienced by the driver ( $a_{yy}$ ) and the time rate of change of the steering wheel ( $\dot{\delta}_{fc}$ ) are kept below reasonable levels during the maneuver. The acceleration and steering rate terms are related to the driver's comfort and workload, respectively. They are "normalized" using  $a_{yyN} = 0.18g$ 's and  $\dot{\delta}_{fcN} = 135 \text{ deg/sec}$  (divided by the steering ratio  $N_S$ ). The driver control  $\delta_{\text{driver}}$  is to be optimally determined to achieve the best tradeoff between these conflicting requirements.

For a given  $W_T$ , the control  $\delta_{\text{driver}}$ , which minimizes  $J$ , can be numerically determined using, for example, the Combined Parameter and Function Optimization Algorithm (CPFA) described in . 13. The optimal steering command for a 2WS vehicle,

obtained with  $W_T = 15$ ,  $U = 60$  km/h, and  $D = -3.6$  m is depicted in Fig. 7. The selected values of  $U$  and  $D$  reflect the conditions of the actual road test. The selected value of  $W_T$  leads to a lane change maneuver time of 1.9 seconds, very close to the time found in the actual road test: delay time = 0.4 seconds, and end time = 2.3 seconds (Fig. 6).

With reference to Fig. 7, the computed optimal steering command compares qualitatively very well with that found with experienced drivers in road tests. Only a qualitative comparison should be made here because the vehicle used in the road test is not the same as that used in our study (lack of data on both the vehicle and the tires used in Ref. 6 prevented us from using that car in our study). These vehicles have different steering sensitivities and steering ratios. As such, the initial steering wheel excursion made in the road test, on the order of 200 degrees, is more than double that found here. Otherwise, these steering commands closely resemble one another.

## Results and Discussions

This “optimal control” approach can thus be used to study and compare the performance of an experienced driver in executing a lane change maneuver using either a 2WS or 4WS vehicle. Results obtained for a 2WS vehicle, at a highway speed of 120 km/h and  $D = -3.6$  m, are compared with those found with 4WSN and 4WSY vehicles (Figs. 8 and 9, respectively). In Fig. 8, we note that the vehicle’s trajectory obtained with the 4WSN vehicle is comparable with that found with the 2WS vehicle. Throughout the maneuver, the heading angle of the 4WSN vehicle is slightly lower than its 2WS counterpart due to the improved damping in the 4WSN’s yaw mode. *However*, in-phase steering of the rear wheels causes an increase in the 4WSN vehicle’s understeer coefficient (or a drop in the steering sensitivity). Hence, larger steering commands were used by the 4WSN vehicle relative to those used with a 2WS vehicle. The increased steering command and steering rate cause an undesirable increase in the driver workload. In contrast, we note in Fig. 9 that the vehicle’s steering command obtained with the 4WSY vehicle is lower than its 2WS vehicle’s counterpart, producing a desirable reduction in the driver workload.

The magnitudes of the cost functional  $J$ , lane change time  $T$ , and other vehicle conditions, for the 2WS, 4WSN, and 4WSY vehicles, are tabulated in Table 2. Note

that the magnitudes of the cost functional obtained with the 4WSN and 4WSY vehicles are smaller than that of the 2WS vehicle, but the differences are not significant. The lane change time and the Root-Mean-Square (RMS) values of the vehicle's lateral acceleration ( $\bar{a}_{yy}$ ) obtained with these three vehicles are very close to one another. The smaller cost functional associated with the 4WSN vehicles come from its smaller heading angle and tire angle found at the end of the maneuver. On the other hand, the cost functional of the 4WSY vehicle comes from its smaller tire angle at the end time as well as the lower transient steering rate. The smaller tire angles achieved with both the 4WSN and 4WSY vehicles will make it easier for the driver to nullify them in the "regulatory" phase of the lane change maneuver

**Table 2 Relative Performance of Vehicles in Lane Change Maneuvers**

Vehicle	$J$	$T$ (sec)	$\psi(T)$ (deg)	$\delta_f(T)$ (deg)	$\bar{a}_{yy}$ (g's)	$\dot{\delta}_{fc}$ (deg/s)
2WS	29.57	1.48	5.2	0.34	0.37	162
4WSN	29.33	1.48	4.9	0.06	0.37	171
4WSY	29.40	1.45	5.5	0.11	0.38	134

Surprisingly, our study indicates that the performance benefit achievable with four-wheel-steering vehicles (using either the 4WSN or 4WSY vehicles) in high-speed lane change maneuvers is not significant for experienced drivers. This is a surprising to us because previous research has concluded that driver/4WS vehicle systems (with either an open-loop or a closed-loop control law) performed better than driver/2WS vehicle systems in collision avoidance maneuvers.<sup>8</sup> However, this favorable conclusion which is derived from a simulation study, is not shared by the conclusions of other simulation studies<sup>17</sup> or "on the road."

In comparing road-test results obtained with both the 2WS and 4WS of a 1987 production vehicle model, the editors of *Road & Track* commented: "... The 4WS does not make itself apparent until the car is pushed quite hard, above, say, 0.8 g's. ...the difference becomes obvious in the slalom. The 4WS is more than 1.0 mph faster."<sup>14</sup> That is, the slalom speeds achievable with the 4WS and 2WS vehicles are 65.5 and 64.5 mph, respectively, a rather small difference. The editors of *Car & Driver* made the following comments on the same car: "...In two days of over-the-road experience with both the two-wheel-steering and four-wheel-steering vehicles, two *Car & Driver* editors

simply could not detect any handling differences between them. " <sup>15</sup> Similarly, comments made on another 4WS production vehicle model are: "... At speed, lane-change maneuvers don't feel different enough to get your attention." ] <sup>6</sup> (However, the author noted the improved performance that that vehicle model offered in fast cornering. ) Results obtained from our study are therefore consistent with those obtained from road tests.

From Table 1, we observe that the response time and damping characteristics of the 4WS vehicles are better than those of the 2WS vehicles. Clearly, our study indicates that "improved" open-loop performance of 4WS vehicles does not necessarily lead to better lane change performance for experienced drivers. This mediocre correlation between the open-loop data and the performance of driver/vehicle systems in lane change maneuvers has also been observed in Ref. 7. Therefore, in designing control algorithms for 4WS vehicles, one should not rely completely on open-loop performance analysis. Instead, promising algorithms from open-loop analyses should be iterated/modified in a "driver-in-the-loop" environment. This can be done either via road tests or using a driving simulator.<sup>7</sup>

## Conclusions

The steering command used by an *experienced* driver during the reflexive phase of a high-speed lane change maneuver has been found to be comparable to that obtained via solving a suitably formulated optimal control problem. This finding allows us to compare the performance of an experienced driver in making a lane change using either a 2WS or 4WS vehicle. For a representative high-speed lane change maneuvers, our study revealed that, in the hands of an experienced driver, the performance benefit achievable with four-wheel-steering vehicles (using either an open-loop or a closed-loop control algorithm) is not significant relative to that with a two-wheel-steering vehicle. This conclusion confirms road test results obtained on two production four-wheel-steering vehicles. The consistency between the road test results and the conclusion obtained from the "optimal control" approach indicates the potential of the proposed methodology as a tool in evaluating the performance of driver/4WS designs in safety-related maneuvers.

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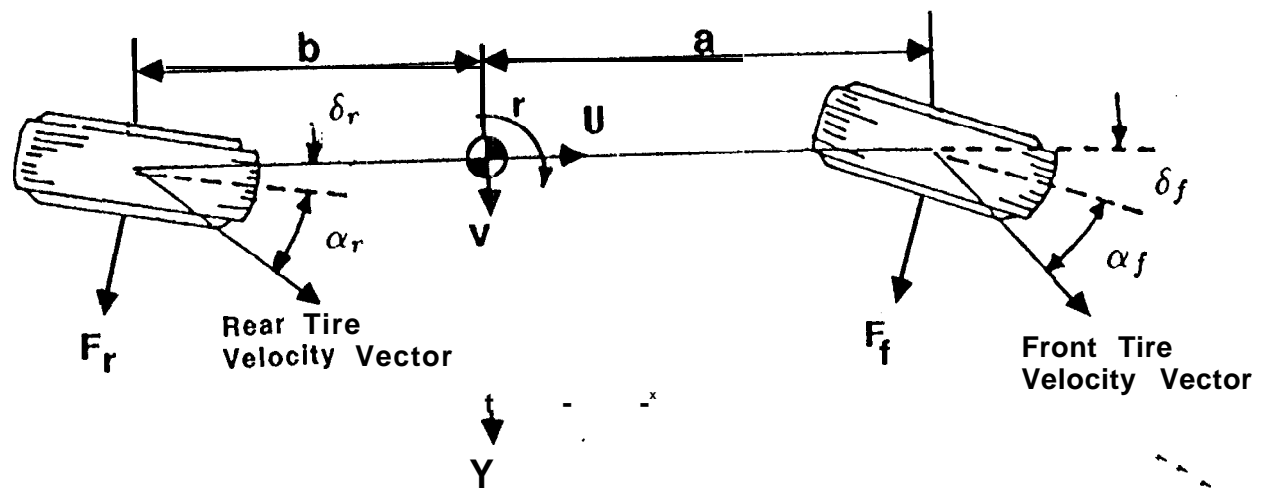


Fig. 1 Schematic of A Vehicle Handling Model



Fig. 2 Variations of lateral acceleration gain with vehicle speed

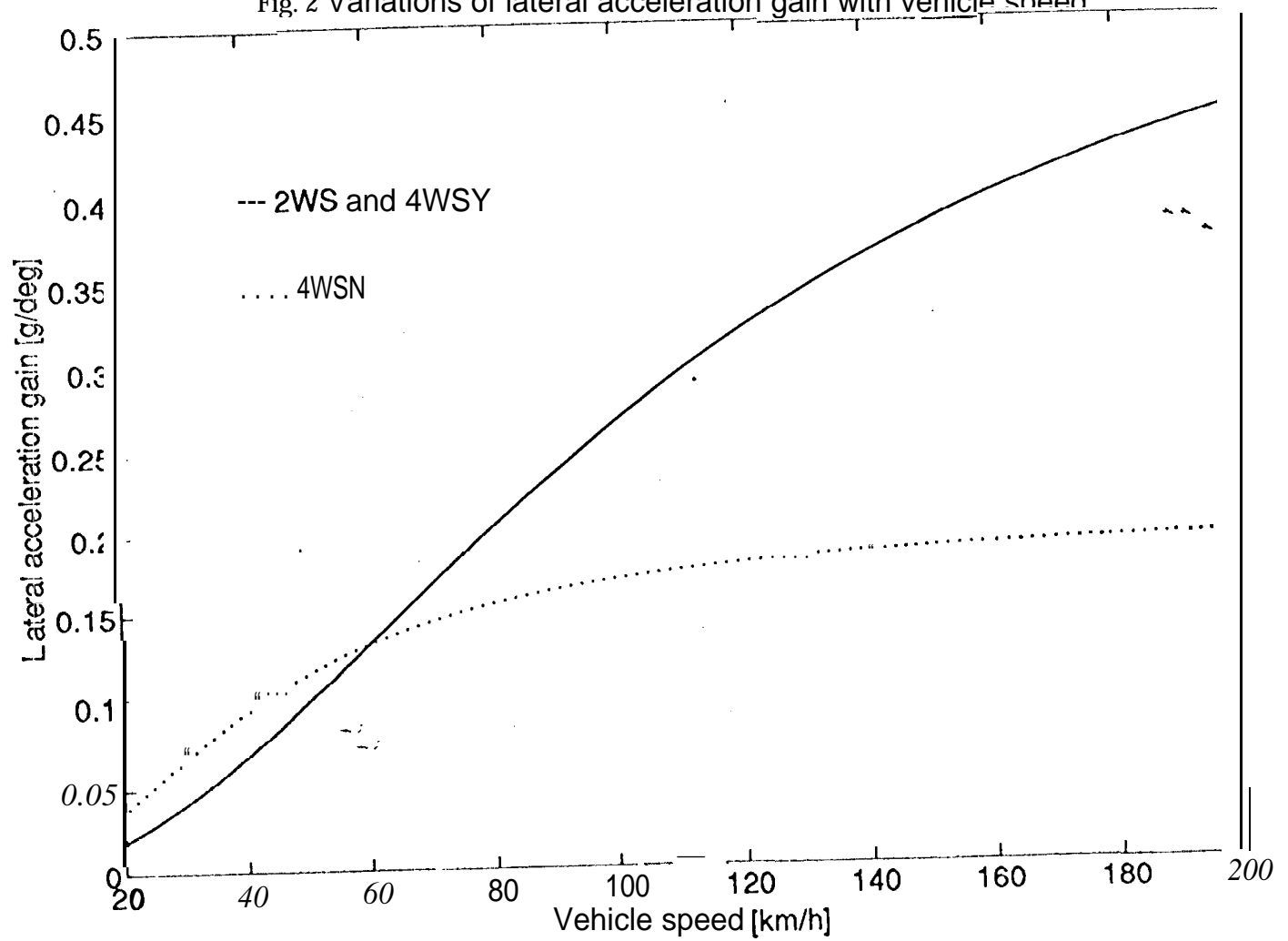


Fig. 3 Variations of yaw velocity gain with vehicle speed

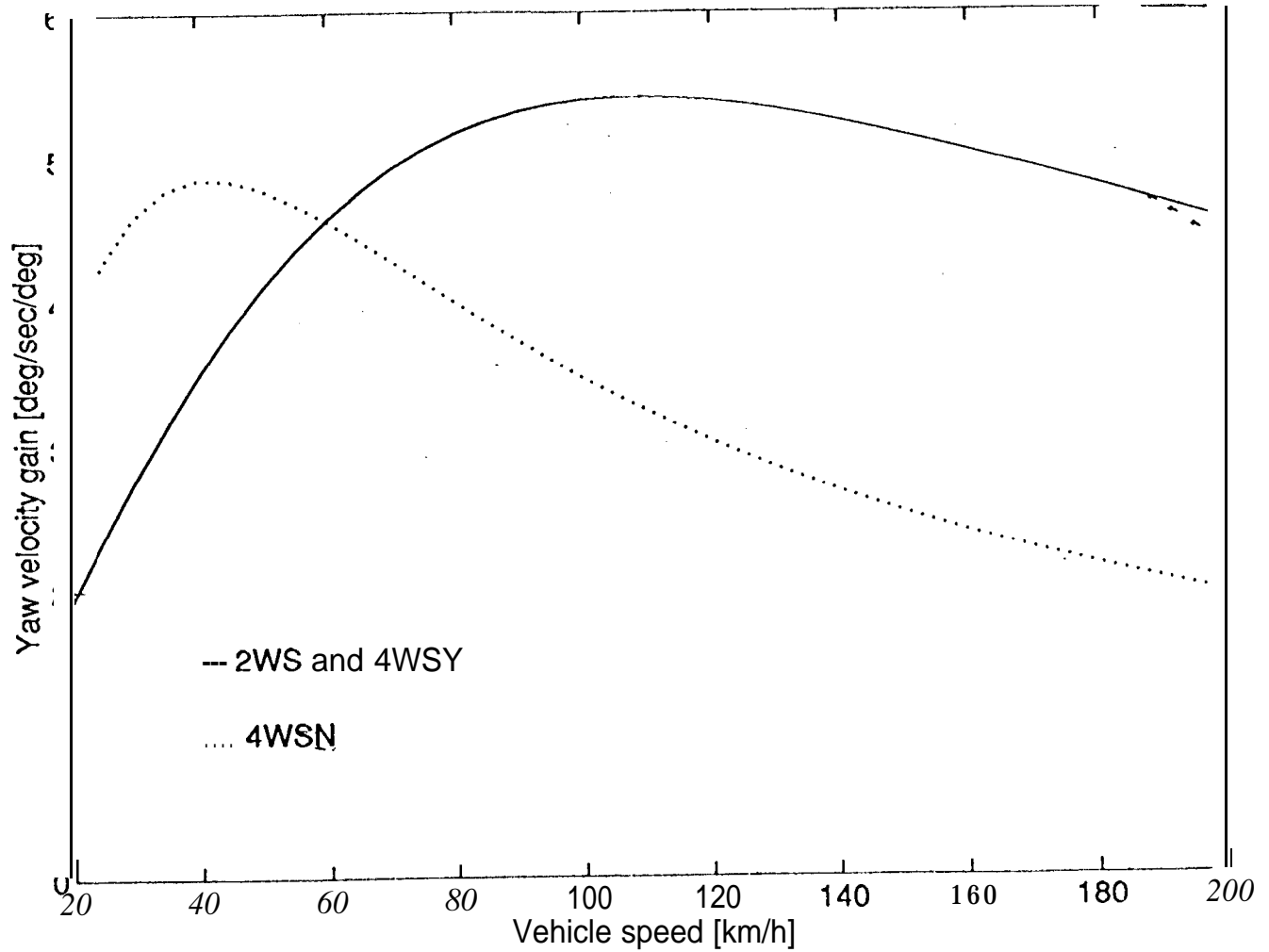
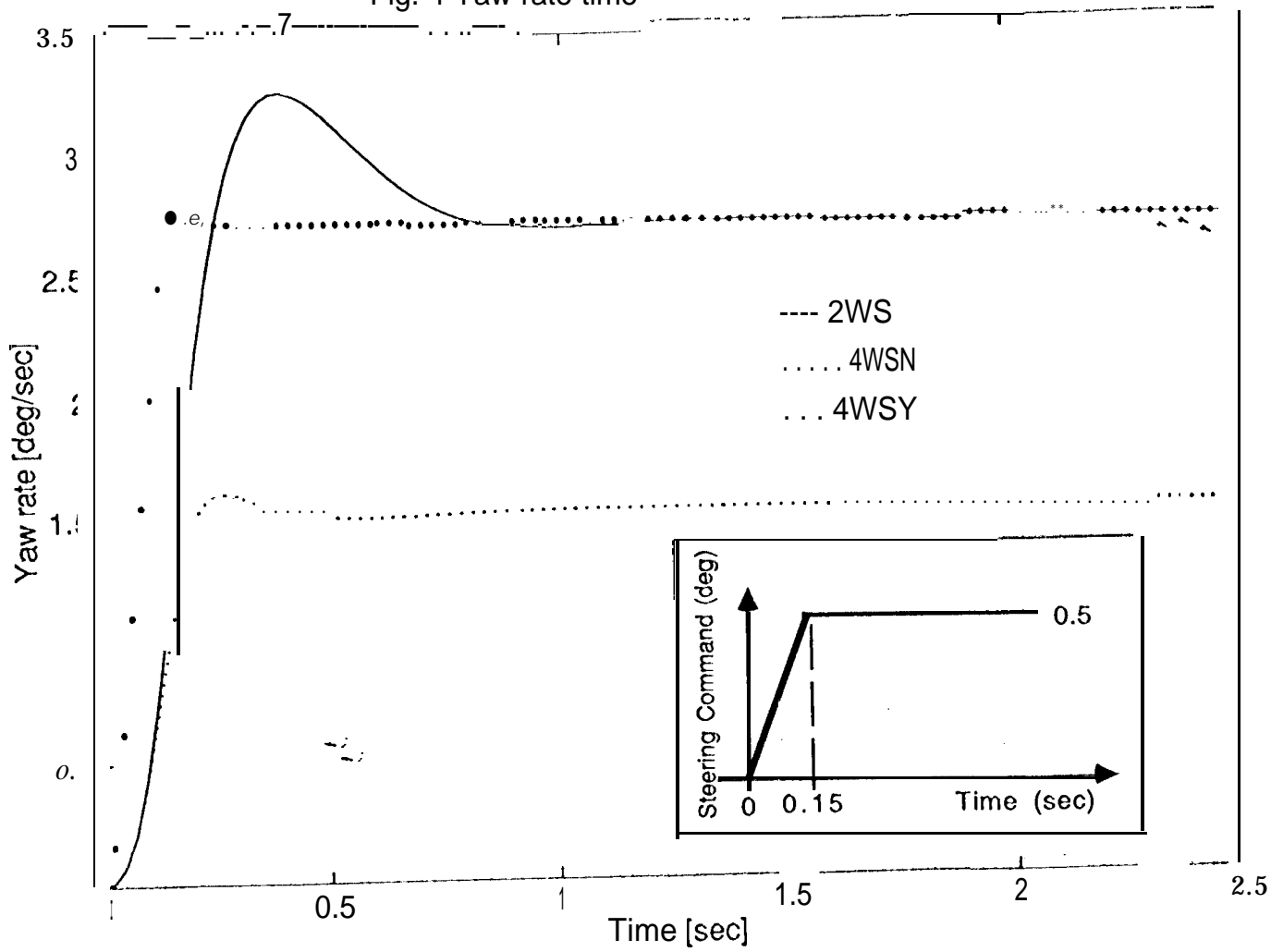


Fig. 4 Yaw rate time responses at 120 km/h



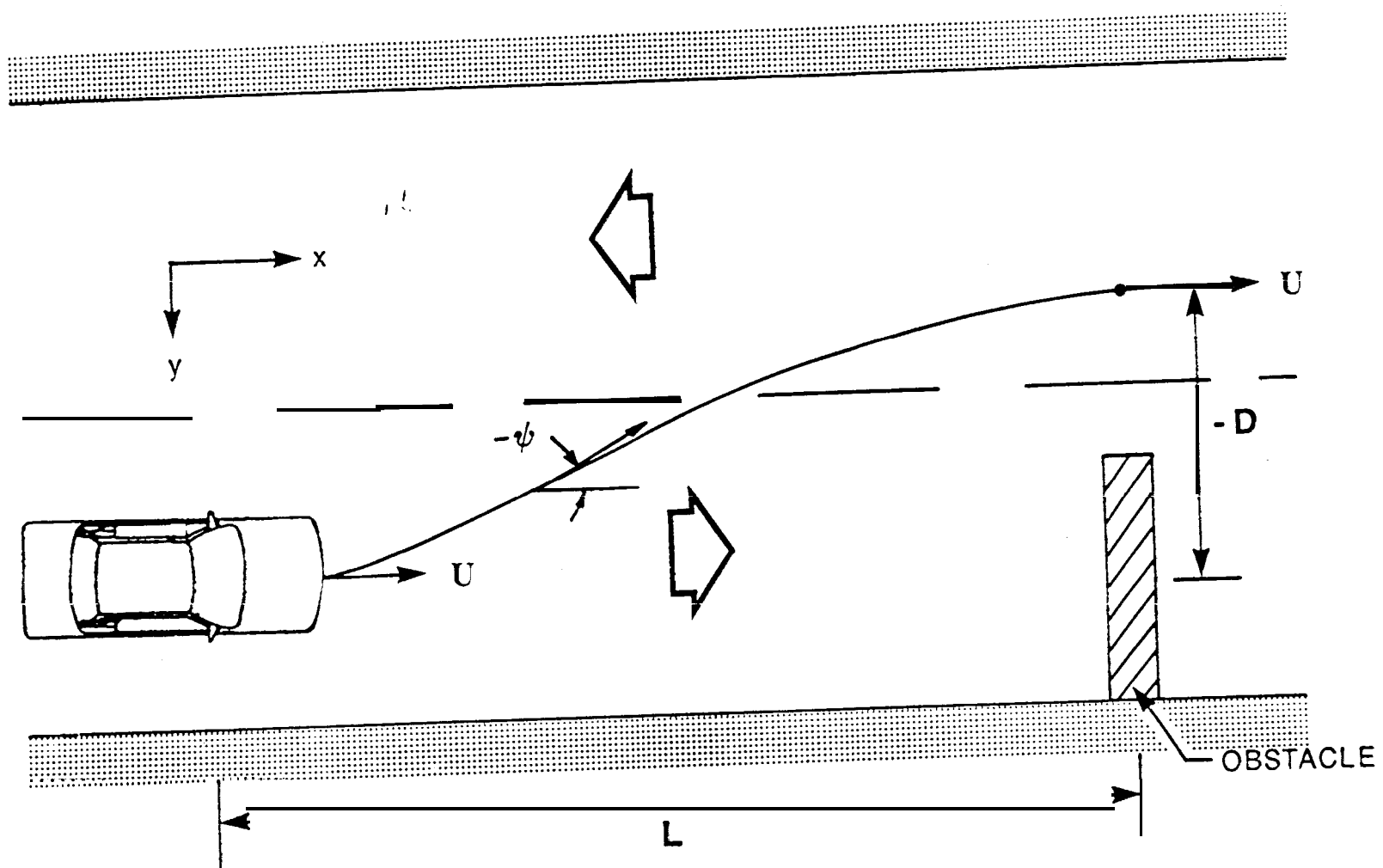
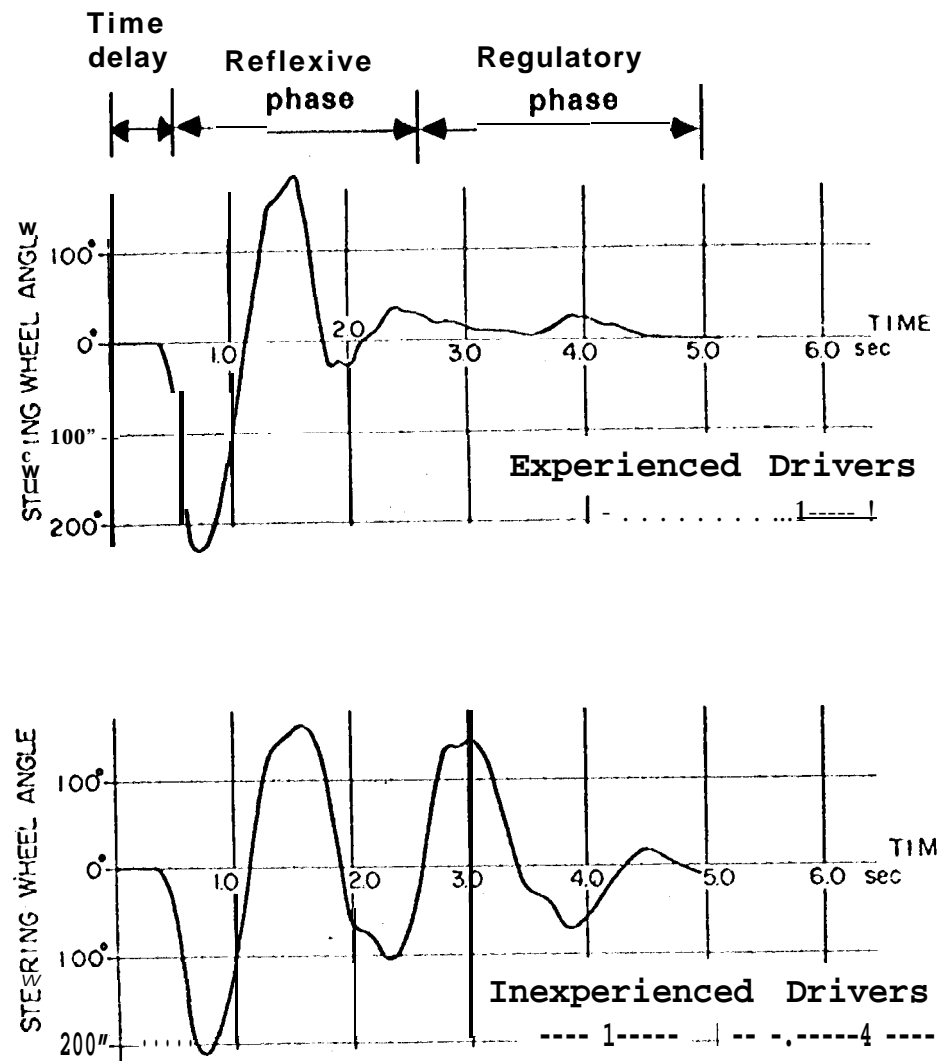


Fig. 5 Schematic of A Lane Change Maneuver



**Fig. 6 Steering Time Histories in A Lane Change Maneuver (Test Results at 60 km/h) [6].**

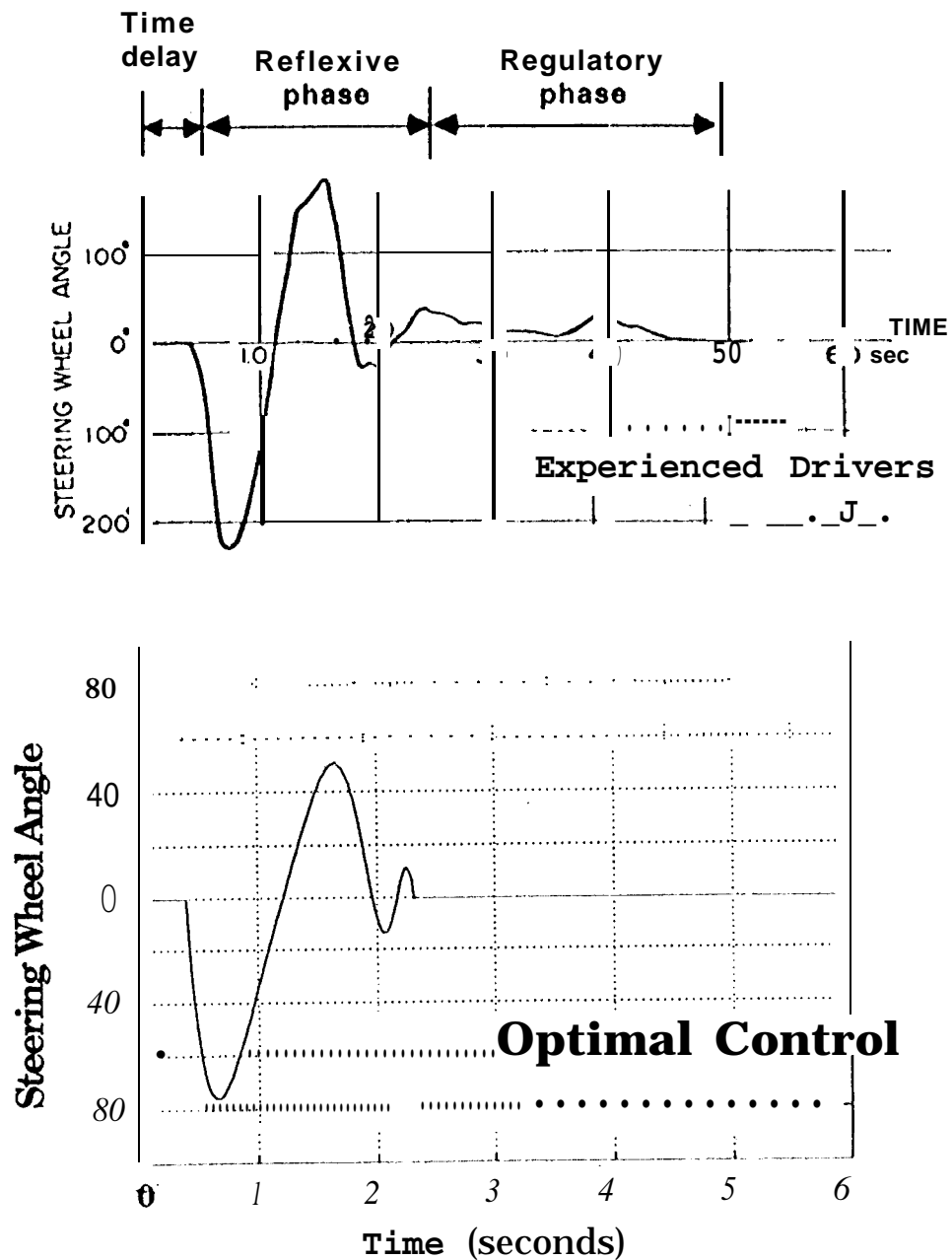


Fig. 7 A Qualitative Comparison of Steering Time Histories (Experienced drivers vs Optimal control). .

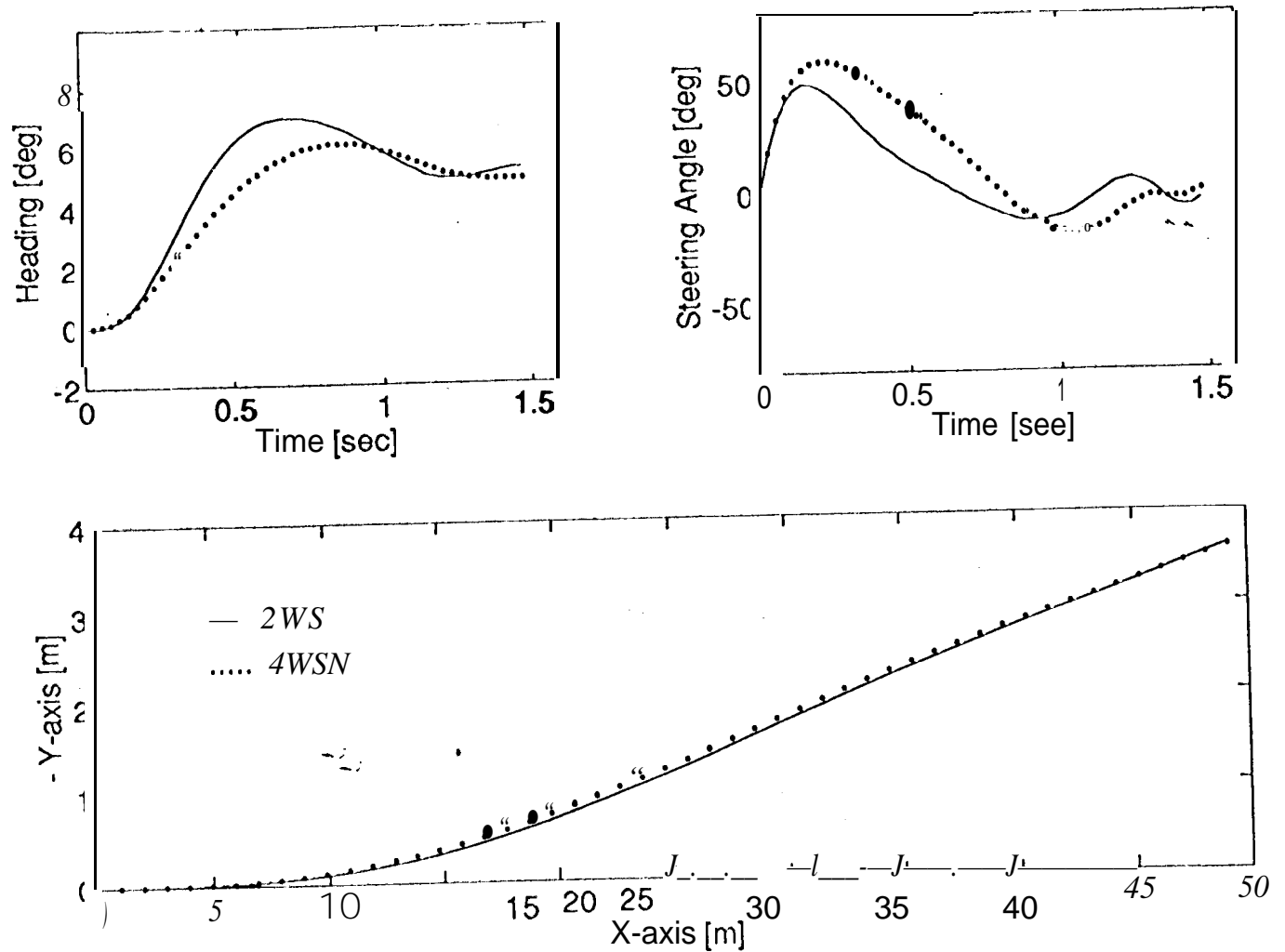


Fig. 8 Comparison of lane change maneuvers (2WS vs 4WSN)

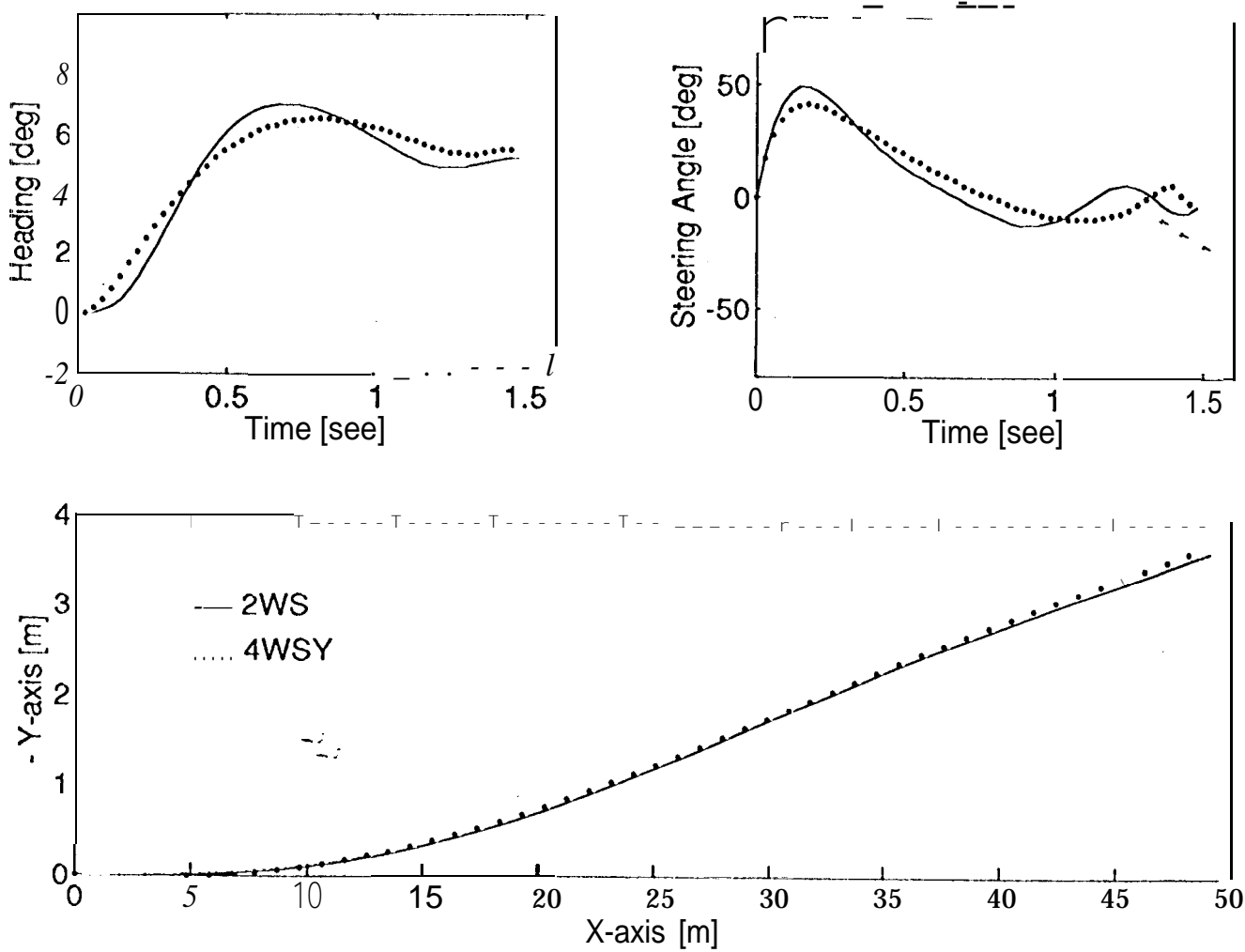


Fig. 9 Comparison of lane change maneuvers (2WS vs 4WSY)